NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL NOTE 2118

INVESTIGATION OF A NACA HIGH-SPEED OPTICAL TORQUEMETER

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SUMMARY

An optical torquemeter the indicating system of which requires no physical connection to the torsion shaft was constructed, and the performance under static and dynamic conditions investigated. Dynamic tests of this torquemeter were made over a range of shaft speeds and torque loads from 6000 to 17,000 rpm and 200 to 5500 inch-pounds. Effects of speed and temperature on the operation of the torquemeter were investigated. Results of static calibration of the instrument are presented.

The magnitude of the speed effects was experimentally determined and applied as a correction factor to the static-calibration equation. The magnitude of the temperature effects was estimated and also applied as a correction factor to the static-calibration equation. The dynamic operation of the instrument over the range of speed and load indicated an over-all root-mean-square accuracy and precision of ±0.51 percent of full load (6300 in.-lb).

Although the accuracy and the precision of this instrument are acceptable, from a consideration of the relative merits of this torquemeter and the strain-gage torquemeter previously investigated the strain-gage torquemeter is better suited to meet the needs of compressor and turbine research.

INTRODUCTION

With the advent of current gas-turbine engines, the problem of obtaining an accurate measurement of shaft torque at high rotative speeds is acute. This need is particularly vital in basic compressor and turbine research, where torque measurements accurate to within ±0.5 percent at shaft speeds up to 17,000 rpm are required.

A torque-measuring instrument inherently having a high degree of accuracy and employing a simple type of indicating system is considered to be the most logical solution to the problem. The performance of one type of torquemeter for this application, based on the measurement of surface strain, is reported in reference 1. Another type of torquemeter that is inherently accurate is the angular-twist type, which utilizes the familiar principle of measuring the twist in a gage length of shaft; the angular twist is proportional to the torque transmitted. The angle may be measured in a number of ways; however, the use of an optical measuring system requires no physical connection to the rotating shaft, and is therefore most desirable.

An optical torquemeter incorporating these basic ideas was designed and built at the NACA Lewis laboratory and its performance evaluated. The optical system used minimized effects of axial and radial movement of the torsion shaft. Effects of centrifugal force at shaft speeds up to 22,000 rpm were evaluated; effects of temperature were approximately evaluated. Results of the performance investigation are presented for a range of shaft speeds from 6000 to 17,000 rpm and torque from 200 to 5500 inch-pounds. Precision and accuracy were experimentally determined in terms of a root-mean-square deviation. Dynamic operation of the instrument is discussed.

APPARATUS

Torquemeter

The optical torquemeter consists of two basic components, a torsion-shaft assembly and an optical indicating system.

Torsion-shaft assembly. - The torsion-shaft (fig. 1) consists of a torsion shaft, two mirror rings containing three mirror surfaces, a sleeve, an oil slinger ring (not shown), and spherical spline couplings. The hollow torsion-shaft is fabricated from SAE 4140 steel, heat treated to a Rockwell hardness of C-26 to C-32. The torsion shaft has an over-all length of 13.54 inches and the gage length of the test section is 9.00 inches. The inside and outside diameters of the torsion shaft are 1.249 and 1.490 inches, respectively. The ends of the shaft have an outside diameter of 2.500 inches. Provisions were made for bolting the spline couplings to each end.

A steel ring (ring A, fig. 1(a)), containing two of the mirror surfaces, was fabricated and rigidly fastened to one end of the torsion shaft as shown. A second steel ring (ring B, fig. 1(a)), containing the third mirror surface, was fabricated and rigidly mounted on a thin steel sleeve. The torsion shaft with ring A attached was placed inside the sleeve, the mirrors oriented, and the sleeve tack

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welded to the torsion shaft at the end opposite the mirror rings on the 2.500-inch-diameter flange. In this manner, a twist in the torsion shaft produced an angular displacement between mirror rings A and B. The final assembly of the torsion shaft after the oil slinger ring and spline couplings were added is shown in figure 1(b).

Optical Indicating System

Description. - The optical indicating system consisted of a commercially available collimator with a modified light source (fig. 2). The light source was a $2\frac{1}{2}$-volt light with a single straight filament. Light from this source was so reflected through a system of mirrors and lenses that it emerged from the collimator in parallel rays. These rays impinged upon the mirrors of the torsion-shaft assembly and were reflected back into the collimator where they were focused on an internal scale, forming an image of the filament of the light source. The scale was graduated from 0 to 25, the smallest division being 0.2. (A scale reading of 25 represents .025 radians of angular deflection in the torsion-shaft assembly.) A magnifying eyepiece for reading the scale completed the assembly. For reading the collimator scale under dynamic conditions, a 6-foot periscope replaced the eyepiece. The insert in figure 2 shows a typical static torque reading of the collimator showing the image of the filament of the light source on the scale.

Principle of operation. - The physical law involved in the operation of the torquemeter states that stress is proportional to strain (within the elastic limit). For a given shaft length, the angular deflection is therefore proportional to the torque applied to the shaft. Difficulty arises, however, in the accurate measurement of this angular deflection. The method for determining the angular deflection for the optical torquemeter is described herein.

A schematic diagram showing in detail the orientation of the three mirror surfaces is presented in figure 3. The two mirror surfaces on ring A form an angle of 90° with respect to one another and this angle is bisected by a plane (indicated by the dotted area), which is perpendicular to the axis of the torsion shaft. The line formed by the intersection of the two mirror surfaces lies in the bisecting plane and makes an angle of 45° with respect to a radial line passing through the axis of the torsion shaft.

The mirror surface on ring B is perpendicular to the bisecting plane of the mirrors on ring A and also forms an angle of 45° with respect to the reference radial line.

Thus, the mirror surfaces are in three planes whose lines of intersection are mutually perpendicular. Such a mirror system reflects light in parallel beams regardless of the orientation of the mirror system as a whole. The reflected image is therefore relatively independent of small radial, axial, and rotative movement of the torsion-shaft assembly and of the position of the collimator with respect to the torsion-shaft assembly. 1298

With the collimator in position and no torque load on the torsion shaft, an image of the filament is formed on the internal collimator scale. This position of the image is the zero or no-load collimator scale reading. When a torque load is applied to the torsion shaft, a twist in the torsion shaft which is proportional to the applied torque, results. This twist causes an angular displacement of mirror ring B with respect to mirror ring A, which, in turn, causes the reflected light to be rotated through an angle that is twice the angular displacement of mirror ring B. The reflected light rays enter the collimator and are focused on the internal scale at a point different from the zero reading. The difference in the collimator reading is proportional to the applied torque.

Experimental Equipment

The torquemeter indicates a quantity that is proportional to torque; therefore, static calibration is necessary in order to determine the factor of proportionality. Static calibration was accomplished by use of a suitable static-calibration apparatus consisting of two pedestals, a ball-type bearing, a 30.00-inch moment arm, and calibration weights (fig. 4).

In order to evaluate the dynamic performance of the torquemeter, it is necessary to have a suitable standard with which to compare values of torque as indicated by the torquemeter. This standard consisted of an eddy-current absorption dynamometer with a separate beam scale to measure dynamometer reaction. The dynamometer was rated at 1700 horsepower at 25,000 rpm and was cradled by self-alineing trunnion bearings between two pedestals. A 63.00-inch moment arm, fitted with a viscous type of vibration damper was provided for static loading. A knife edge was bolted to the arm at approximately 35 inches from the axis of the dynamometer and was used to apply both static and dynamic loads to the beam scale. Dynamic loading was accomplished by changing the field current. Heat generated in the stator was removed by circulating water through a stator jacket. Temperature of the cooling water at the outlet was held at 120 ± 20° F.

A 3000-horsepower variable-speed motor with a speed-increasing gearbox was used as a power source.

EXPERIMENTAL SETUP AND METHODS

The general experimental program consisted in evaluating the torquemeter characteristics under static and dynamic conditions. Static calibrations of the torquemeter were performed at room temperatures, approximately 80° F. In order to evaluate the dynamic operative characteristics of the torquemeter, it was necessary to calibrate the dynamometer statically to measure the actual dynamic loads applied to the torquemeter. A no-load spin test was performed to determine the effects of centrifugal force on the mirror rings.

Static Calibration

<u>Dynamometer static calibration.</u> - Static calibration of the dynamometer was accomplished by placing the calibration weights on a loading pan attached to the 63.00-inch station of the dynamometer moment arm and recording the corresponding scale reading. Readings were thus recorded for both increasing and decreasing torque loads in increments of 315 inch-pounds.

Precautions were taken to insure accurate and representative dynamometer calibrations. Water- and oil-hose connections were orientated to minimize any torque application to the dynamometer by forces transmitted through these connections. Dynamometer oil and water pressures were maintained constant and the discharge temperature of the cooling water was approximately 120° F.

Plots of dynamometer scale reading against applied torque were close to a straight line and were expressed in the form of a slope-intercept equation. The method of least squares was employed to determine the particular values of slope and intercept, that gave the best fit to the observed calibration data.

The equation was of the form

$$Y = MX + B \tag{1}$$

where

Y scale reading, pounds

M constant (slope)

X torque, inch-pounds

B constant (intercept)

The results of a typical dynamometer static calibration are presented in figure 5. The error in inch-pounds is plotted against true torque values, which were calculated from the calibration equation determined by the method of least square. The average error of the observations for any true torque value is within 0.1 percent of full-load torque.

The root-mean-square error of any one observed static reading, based on a number of calibrations, is ±0.39 percent of full-load torque. The calibration was repeated at intervals during the course of the torquemeter dynamic-operative studies to correct for the variation of the dynamometer static calibration with time.

Torsion-shaft static calibration. - The torsion-shaft static-calibration apparatus is shown in figure 4. The loading pan was attached to the 30.00-inch static-moment arm by means of knife edges. The moment arm with the loading and the counter-balance pans attached was balanced and keyed rigidly to one end of the torsion shaft. The torsion shaft was mounted between two pedestals, one end being supported by a ball bearing located on the rear pedestal, the other end bolted rigidly to the front pedestal. After mounting, the moment arm and the torsion shaft were adjusted to a horizontal position and the moment arm set perpendicular to the axis of the torsion shaft. The collimator was so placed that an image was formed on its internal scale; the light source was so adjusted that a zero or no-torque reading was obtained. Each of the specially constructed calibration weights were 5 ± 0.004 pounds. Static calibrations were made by placing the calibration weights on the loading pan and recording the corresponding collimator reading. Readings were thus recorded for both increasing and decreasing torque from 0 to 6300 inch-pounds in increments of 150 inch-pounds.

Dynamic Operation

Dynamic operation of the torquemeter was divided into two phases: the first phase consisted in dynamic loading of the instrument over a range of torque and speed; the second phase was a no-load spin test to determine the effects of centrifugal force on the instrument reading.

Dynamic load tests. - For dynamic operation under varying-load conditions, the torquemeter was coupled between the high-speed pinion drive shaft of the gearbox and the dynamometer drive shaft as schematically shown in figure 6. Coupling was made through spherical-type splines, and sufficient axial clearance was provided in the couplings to prevent any axial stress in the torsion shaft. The oil for spline-coupling lubrication entered the torsion shaft from the hollow high-speed pinion of the gearbox. The baffling and scavenging system was necessary in order to keep any oil mist from contaminating the mirrors of the torsion shaft.

The 3000-horsepower variable-speed motor in conjunction with the speed-increasing gearbox was used as a power source. Power thus generated was transmitted through the torsion shaft and absorbed by the dynamometer. Simultaneous values of torque were indicated by the torquemeter and dynamometer reaction.

Dynamic load runs were made over a range of speeds from 6000 to 17,000 rpm. Critical vibration of the dynamometer prevented sustained operation from 9000 to 14,000 rpm. Torque loads from 200 to 5400 inch-pounds were set over the range of speed from 6000 to 9000 rpm. At higher speeds, the maximum load was reduced to 4500 inch-pounds. Runs were made by maintaining constant speed while the load was varied. Simultaneous readings of the torquemeter and dynamometer reaction were recorded during these runs.

Spin test. - The torquemeter was spun from 4000 to 22,000 rpm at no load in order to determine the effect of centrifugal force on the torquemeter reading. The additional parts that were added to the torsion shaft in order to perform the no-load spin run are shown in figure 7(a). An adaptor shaft, which carried the friction load of the outboard bearing, fastened the male spline coupling into place and passed through the hollow torsion shaft. A centering bushing made an easy sliding fit over the adaptor shaft and fastened into the end of the torsion shaft. An outboard ball-type bearing completed the assembly. The final assembly of the torsion and adaptor shaft is shown in figure 7(b).

The modified torsion-shaft assembly installed on the gearbox is shown in figure 7(c). Provisions for oiling, scavenging, and temperature and vibration measurements were made on the outboard bearing pedestal. The collimator was installed as shown and the torsion-shaft assembly was then spun from 4000 to 22,000 rpm and the change in the zero reading of the collimator recorded.

RESULTS AND DISCUSSION

The evaluation of an instrument is based on the qualities, accuracy, precision, and practicability. Accuracy is defined by the order of agreement between the magnitude of a quantity as indicated by the instrument and the true magnitude. The precision is determined by the ability to repeat a reading under a given set of conditions. The practicability is judged by the efficiency (in terms of reliability, simplicity, convenience, and safety) with which the instrument may be used to achieve the purpose for which it was designed. The evaluation of this torquemeter under static and dynamic conditions is based on these qualities.

Torsion-Shaft Static Calibration

Static calibration is necessary in order to determine the factor of proportionally, which exists between the collimator reading and the applied shaft torque. Such a determination is achieved by static calibration. For static calibration, accuracy was considered to be the order of agreement of any observation with the corresponding true value as established by applying the method of least squares to all the observations.

A typical static calibration of the torquemeter is shown in figure 8(a), where the collimator reading is plotted against true shaft torque in inch-pounds. The plot of the observed points closely approaches a straight line. Accordingly, a linear relation between collimator reading and applied shaft torque was assumed in the form

$$y = mx + b \tag{2}$$

where

y collimator reading

m slope of line

x applied torque, inch-pounds

b value of y at zero torque (intercept)

The method of least squares was employed to determine the particular values of m and b that made equation (2) fit the observed data best. The error of the observed data from the calibration equation thus determined, is shown in figure 8(b). Figure 8(b) shows the hysteresis present in the torsion shaft and the accuracy with which the observed data points indicate the true value of shaft torque. High hysteresis values in the static calibration of the torsion shaft were caused by friction between the sleeve and the torsion shaft. Inasmuch as these rubbing surfaces could not be lubricated because of the proximity of the mirror surfaces, this hysteresis tended to increase with time. In future designs this condition should be remedied.

The root-mean-square error of any individual reading, based on all the observed data for this calibration, is within ±17 inch-pounds or ±0.27 percent of full load (6300 in.-lb).

Because torsion-shaft static calibrations were performed at room temperatures (approximately 80° F), correction factors based on the change in the torsional modulus of elasticity with temperature were applied to determine the proper static calibration for temperatures other than 80° F. The slope of the static calibration is inversely proportional to the torsional modulus of elasticity of the torsion shaft.

When the slope of one calibration is known at a given temperature, it is relatively simple to find the slope of a calibration at any other temperature by the use of the following equation (reference 2, p. 31).

$$\frac{G}{G_0} = 1 - \left(\frac{T}{T_m}\right)^2$$

where

G torsional modulus to be determined

G_0 torsional modulus at 0° R

T temperature at which G is desired, °R

T_m melting point of material, °R

On the basis of experience with shafts operating under similar conditions, shaft temperatures were assumed to be 80° F up to 8000 rpm, 100° F up to 10,000 rpm, and 130° F for shaft speeds over 10,000 rpm. The magnitude of this correction amounted to a 0.6-percent increase in the slope of the static calibration for a torsion-shaft temperature rise of 50° F.

Dynamic Operation

Centrifugal-force effects. - The results of the no-load spin test are presented in figure 9. The change in the zero or no-load collimator reading is plotted against torsion-shaft speed in rpm. No change occurred in the collimator reading for shaft speeds from 4000 to 8000 rpm. From 8000 to 10,000 the reading increased 0.025 divisions, which represents a torque change of 7 inch-pounds or approximately 0.11 percent of a full-load torque of 6300 inch-pounds. At shaft speeds above 10,000 rpm the increase in collimator reading was 0.05 divisions and remained constant until approximately 22,000 rpm, which was the maximum speed for the test, was attained. These centrifugal effects were included as correction factors on the static-calibration equation.

Thus, application of two correction factors to the static-calibration equation is necessary: a temperature correction on the value of the slope m, and a shaft speed correction on the value of the intercept b. The observed collimator reading together with speed and temperature corrections was used in equation (2) to calculate dynamic shaft torque.

Precision and accuracy. - Dynamic shaft torques, as indicated by the dynamometer, were assumed to be true torque values and were used as a basis for evaluating the precision and the accuracy of the torquemeter under dynamic conditions. The difference between simultaneous torque values, as indicated by the torquemeter and dynamometer, is called the torquemeter deviation. This deviation for a range of speed and load is plotted against dynamometer torque in figure 10.

The precision of the torquemeter under dynamic conditions is indicated by the data-point groups for similar conditions of speed and load (fig. 10). Furthermore, because the dynamometer torque values are assumed to be true torque values, the proximity of the data points to the line of zero torquemeter deviation is an indication of the accuracy of the torquemeter.

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In general, the precision and the accuracy of the torquemeter is higher at shaft speeds at and below 14,000 rpm. At higher speeds the greater scatter of the data was caused by torsional vibration making it difficult to obtain accurate collimator readings. The greater deviation may be caused by an error in the assumed shaft temperature. In the range of speed below 14,000 rpm, the normal oscillation of the image on the collimator scale was on the order of ±0.3; whereas at higher shaft speeds, the amplitude of the oscillation was on the order of ±0.6. The root-mean-square deviation based on all the dynamic data points is ±32 inch-pounds or ±0.51 percent of full load (6300 in.-lb) and represents the over-all accuracy and precision. The root-mean-square error in the agreement of torquemeter and dynamometer torques based on the static calibrations is ±29 inch-pounds or ±0.46 percent of full load. This root-mean-square error shows that the probable accuracy and precision of the instrument obtained under dynamic conditions was fairly accurately predicted by applying the laws of probability to the static calibrations of the torquemeter and the dynamometer.

Practicability

The practicability of the instrument may be summarized in the following observations:

(1) The investigation proved that an optical-type torquemeter may be operated at high rotative speeds with considerable accuracy.

(2) The principles involved in its operation are simple and the collimator used with this torquemeter is commercially available.

(3) The installation of the torquemeter is greatly simplified by the use of an optical indicating system that requres no physical connection to the rotating torsion shaft.

Some of the disadvantages associated with the use of the particular optical system employed with this torquemeter are:

(1) Difficulty is encountered in the actual fabrication of mirror rings and mirrors to close tolerances.

(2) A baffling system is necessary to keep oil mist from the surfaces of the mirrors and the lenses.

(3) Failure of the filament of the light source requires recalibration of the torsion shaft because the position of the filament in any two bulbs is not exactly the same.

(4) Remote readings of the instrument are difficult to obtain and even though a 6-foot periscope was used it was necessary for personnel to be near the rotating torsion shaft. This proximity to the shaft is particularly undesirable from a safety view point. 1298

Although the accuracy and the precision of the optical torquemeter are acceptable, from a consideration of the comparative merits of this torquemeter and the strain-gage torquemeter (reference 1), based on experience with the two instruments, the strain-gage torquemeter is better suited to meet the needs of basic compressor and turbine research.

CONCLUDING REMARKS

From an investigation involving a study of the static and dynamic operating characteristics of an optical torquemeter, the following results, conculsions, and recommendations were obtained:

1. An optical torquemeter, the indicating system of which requires no physical connection to the torsion shaft, was developed and operated over a range of speed and load from 6000 to 17,000 rpm and 200 to 5500 inch-pounds with an accuracy and precision of ±0.51 percent of full load (6300 in.-lb).

2. Calibration of the torquemeter was affected by shaft temperature and speed. Measurements of torsion-shaft temperature should be made to permit accurate correction of the static-calibration constants. The magnitude of the effects of centrifugal force can be determined by a suitable no-load spin test.

3. High hysteresis values in static calibration of the torquemeter were apparently caused by friction between the sleeve and the torsion shaft. This condition should be corrected in future designs.

4. On the basis of experience with the optical torquemeter and with the strain-gage torquemeter, in the present form the strain-gage torquemeter is better suited to meet the needs of basic compressor and turbine research.

Lewis Flight Propulsion Laboratory,
National Advisory Committee for Aeronautics,
Cleveland, Ohio, March 7, 1950.

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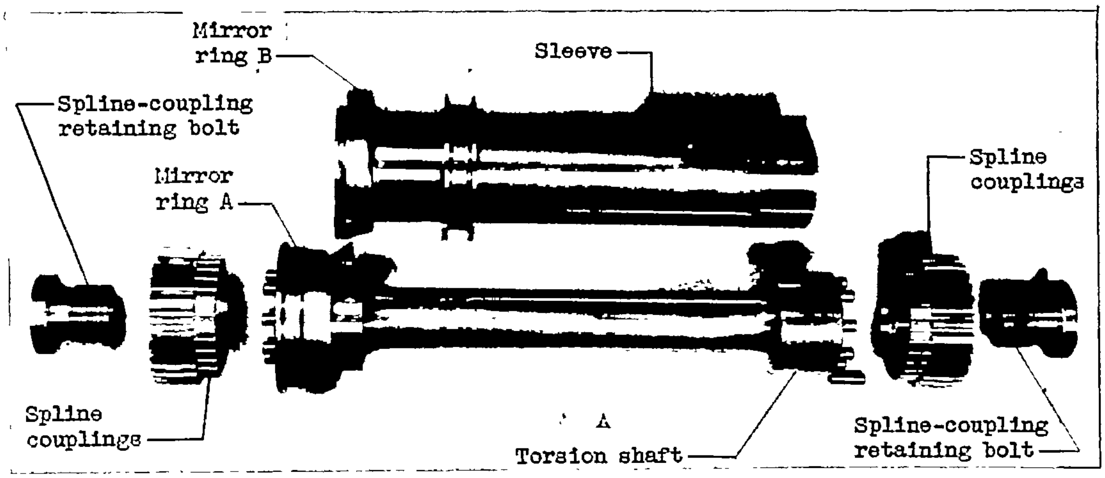
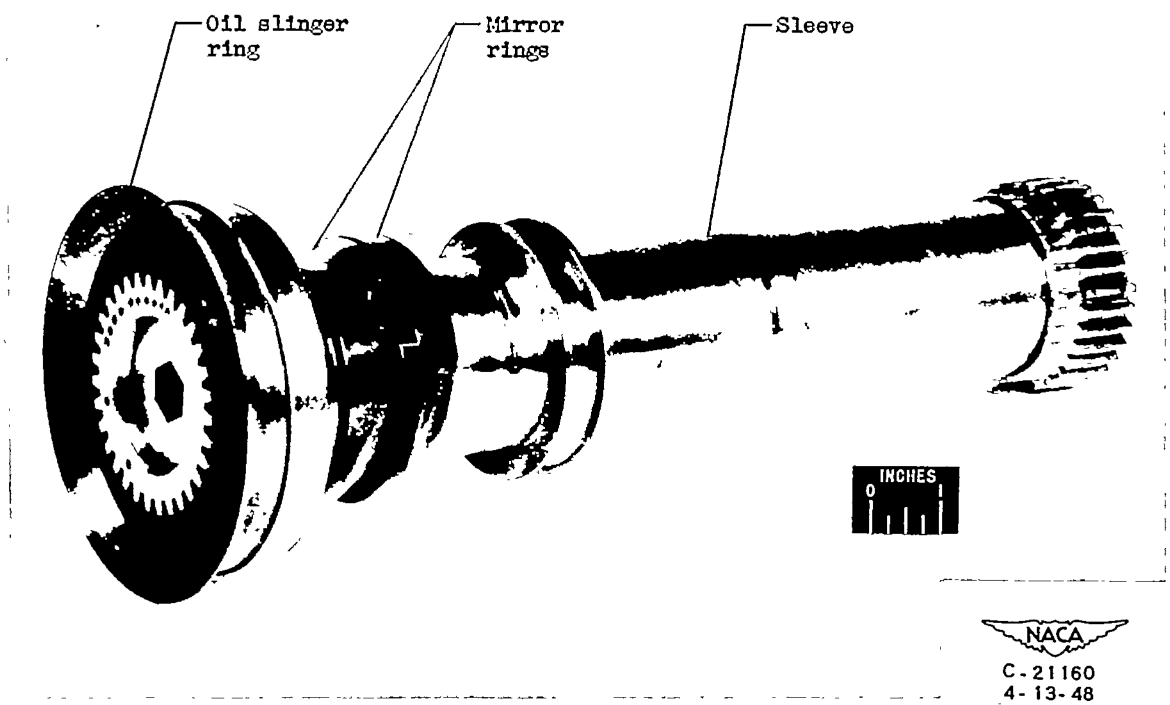
1. Rebeske, John J., Jr.: Investigation of a NACA High-Speed Strain-Gage Torquemeter. NACA TN 2003, 1950.

2. Sutherland, William: A Kinetic Theory of Solids, with an Experimental Introduction. Phil. Mag. and Jour. Sci., vol. 32, no. CXCIV, ser. 5, July 1891, pp. 31-43.



(a) Disassembled.



(b) Assembled.

Figure 1. - Torsion-shaft.

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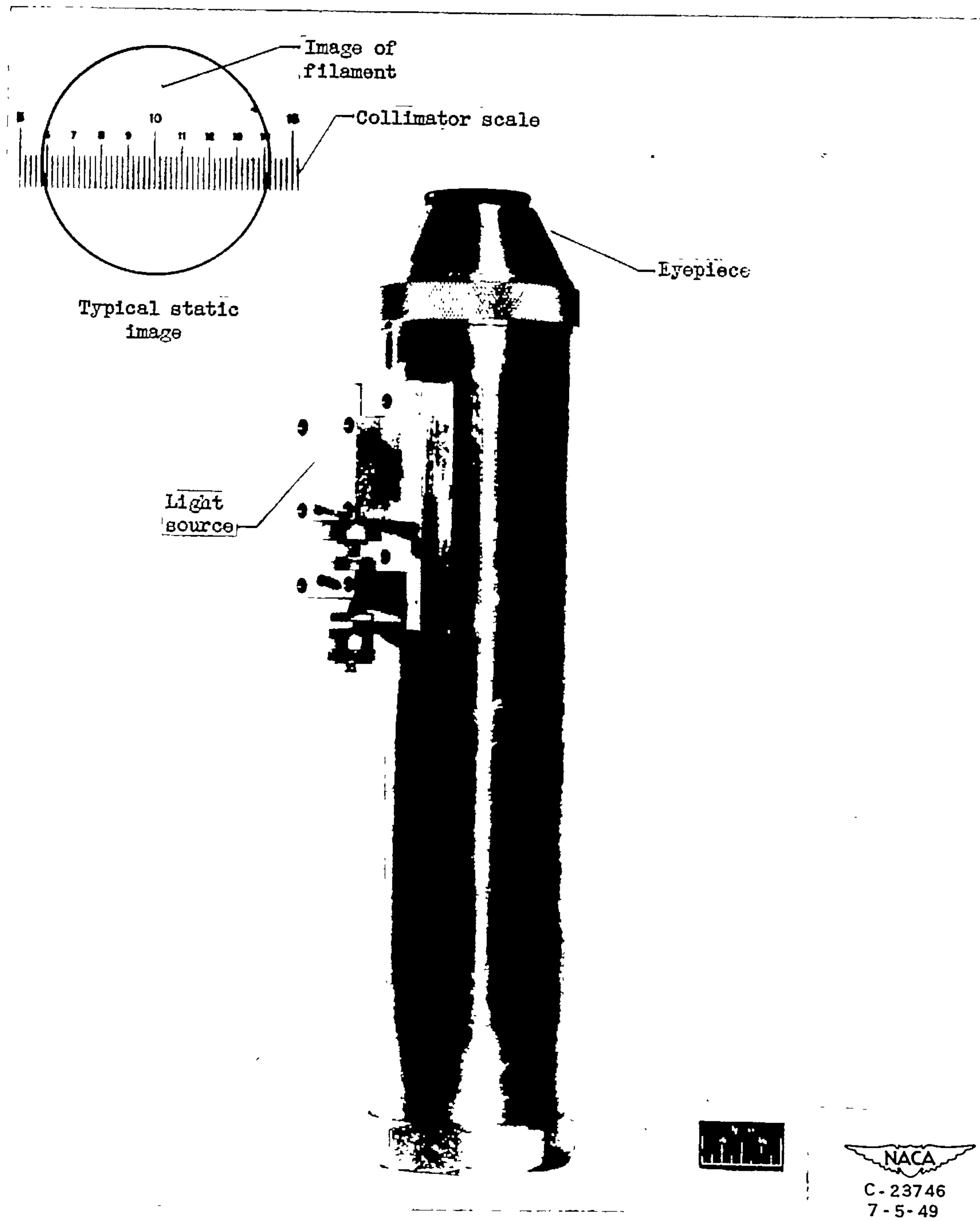

Figure 2. - Optical collimator with insert showing typical torquemeter reading under static conditions.

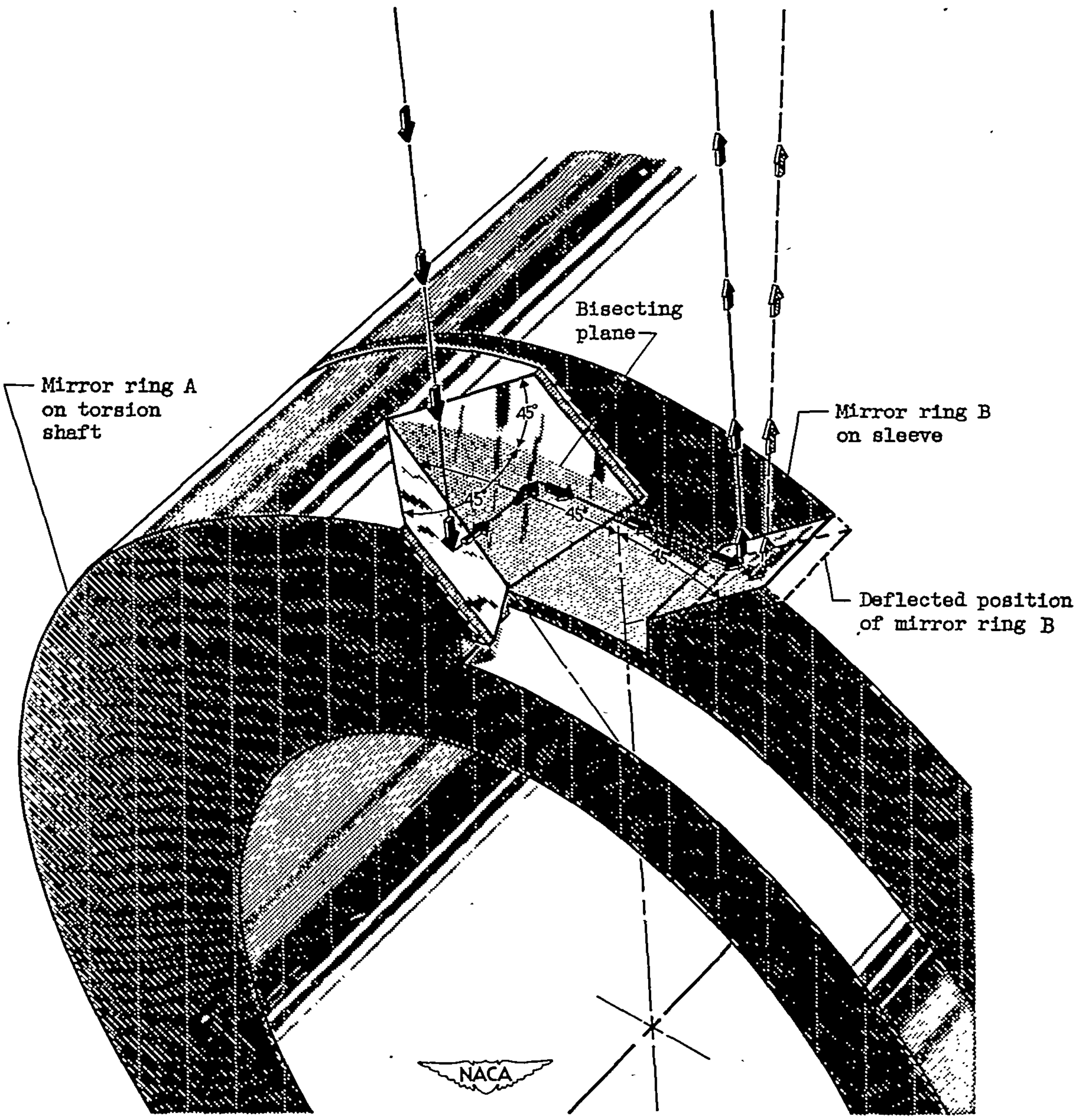

Figure 3. - Schematic diagram of orientation and operation of mirror surfaces of torsion-shaft assembly.

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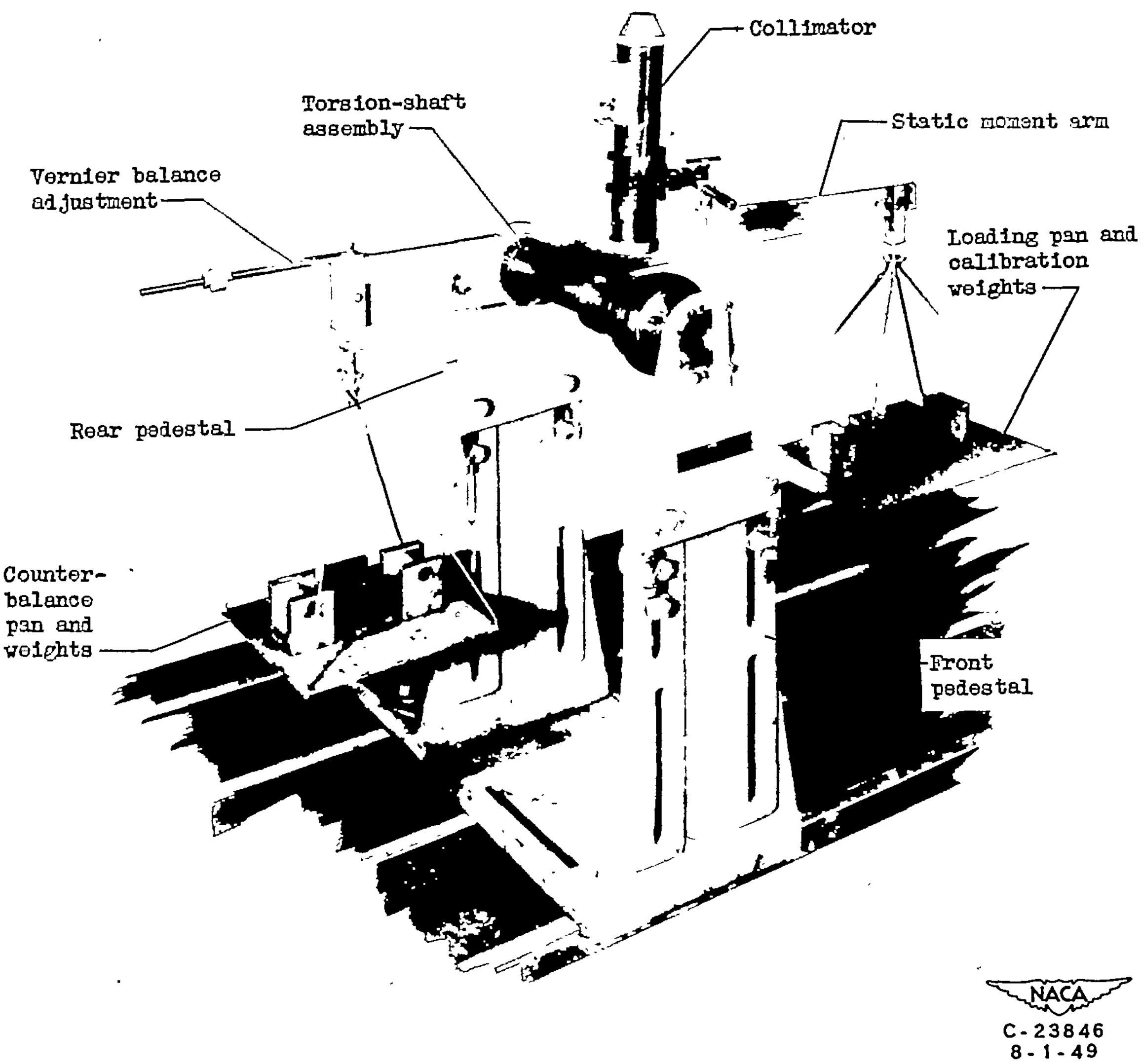

Figure 4. - Static-calibration apparatus for optical torquemeter.

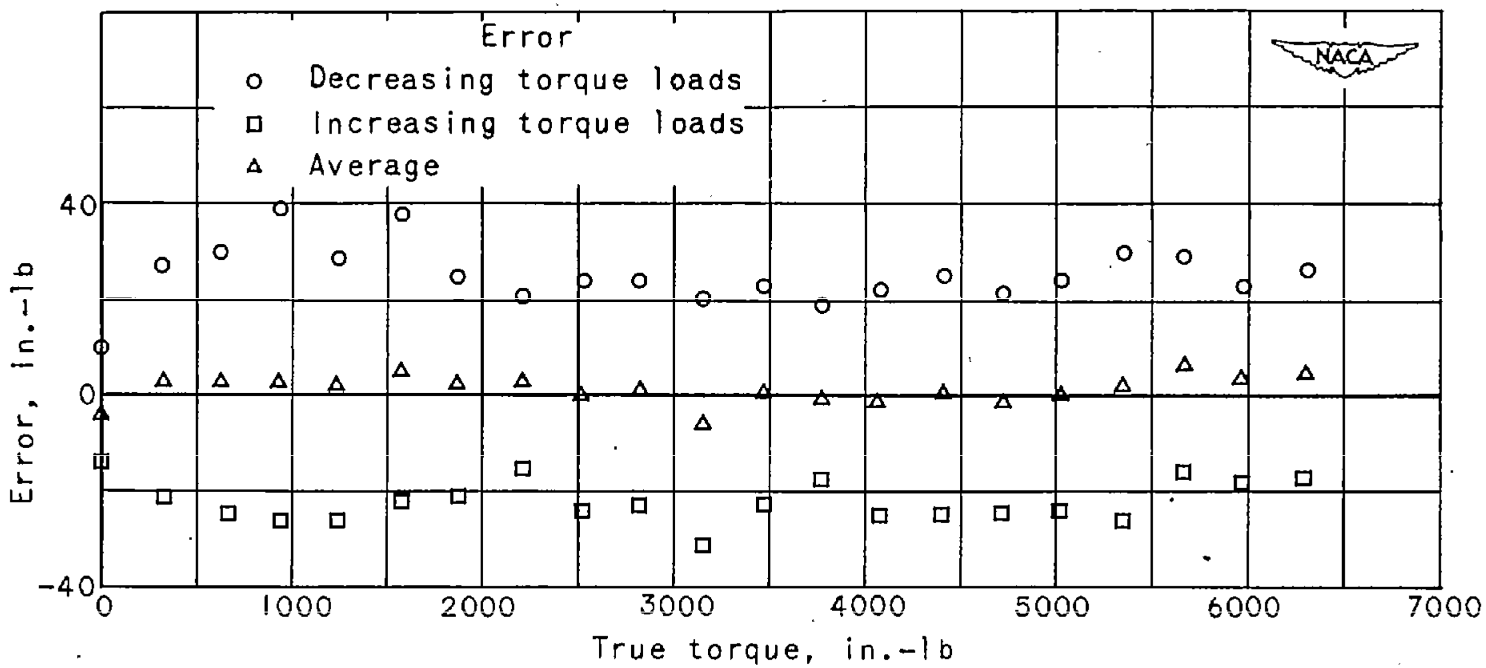

Figure 5. - Results of typical dynamometer static calibration.

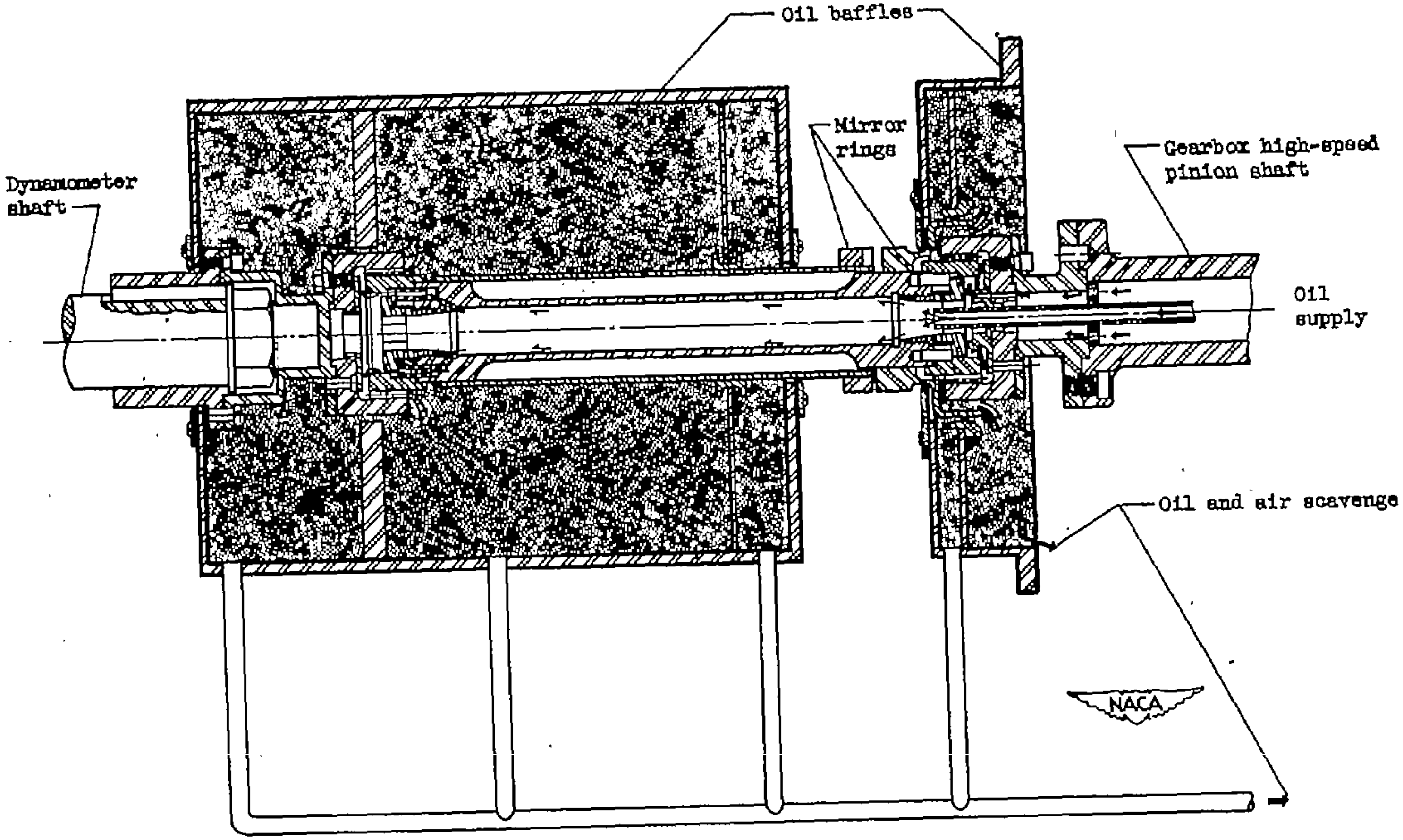

Figure 6. - Installation of torsion shaft for dynamic operation.

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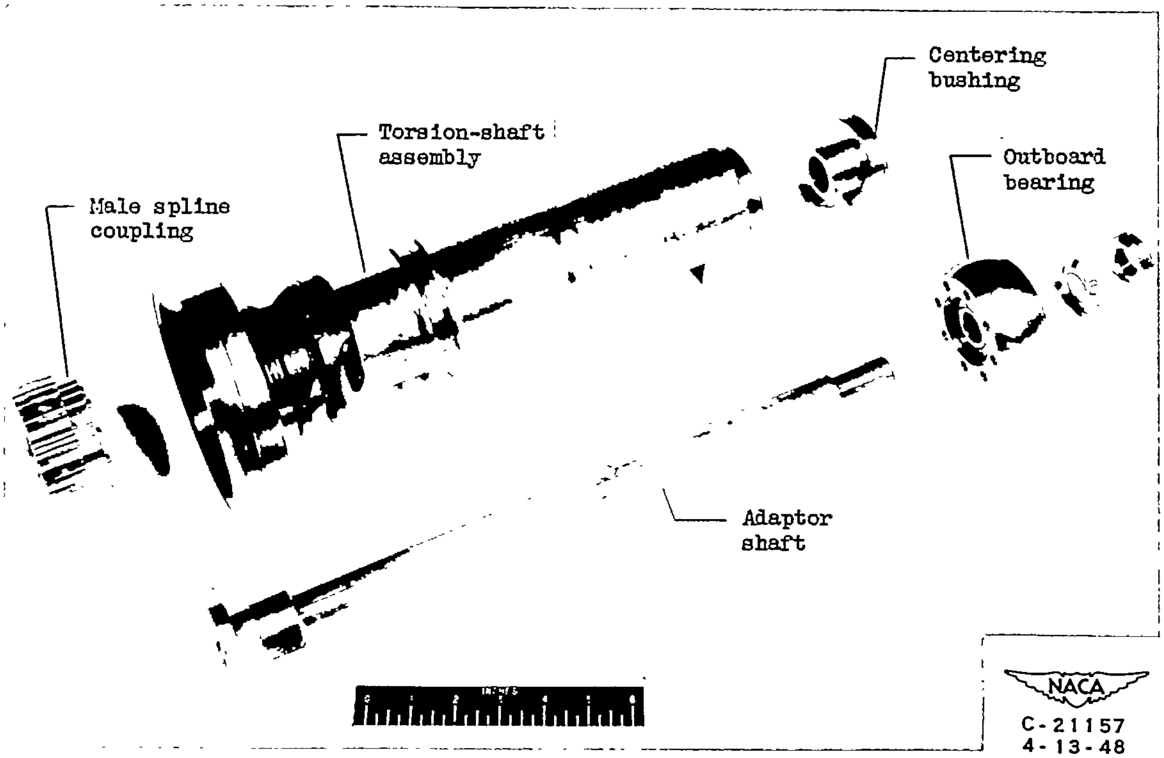
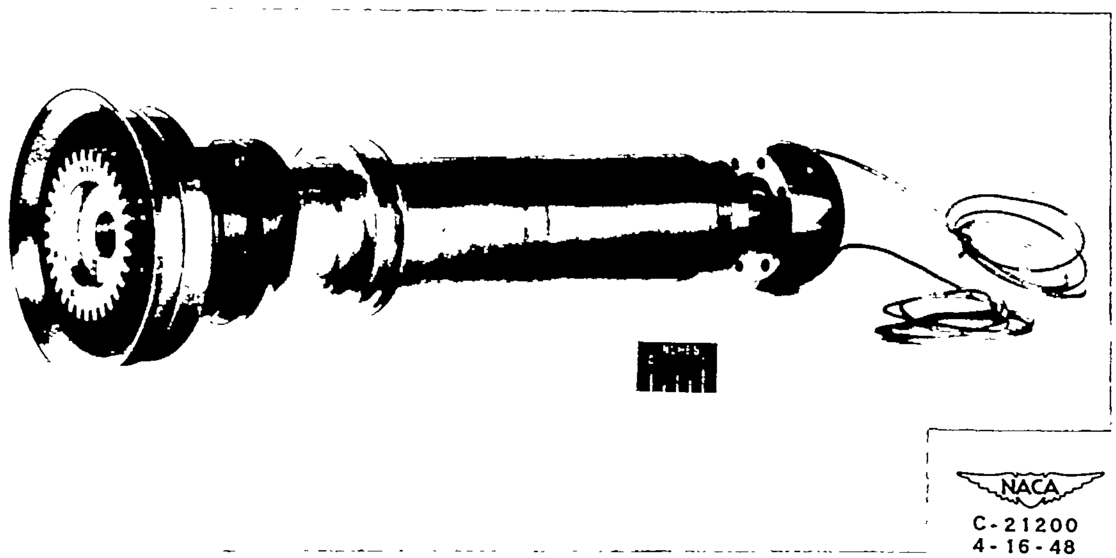
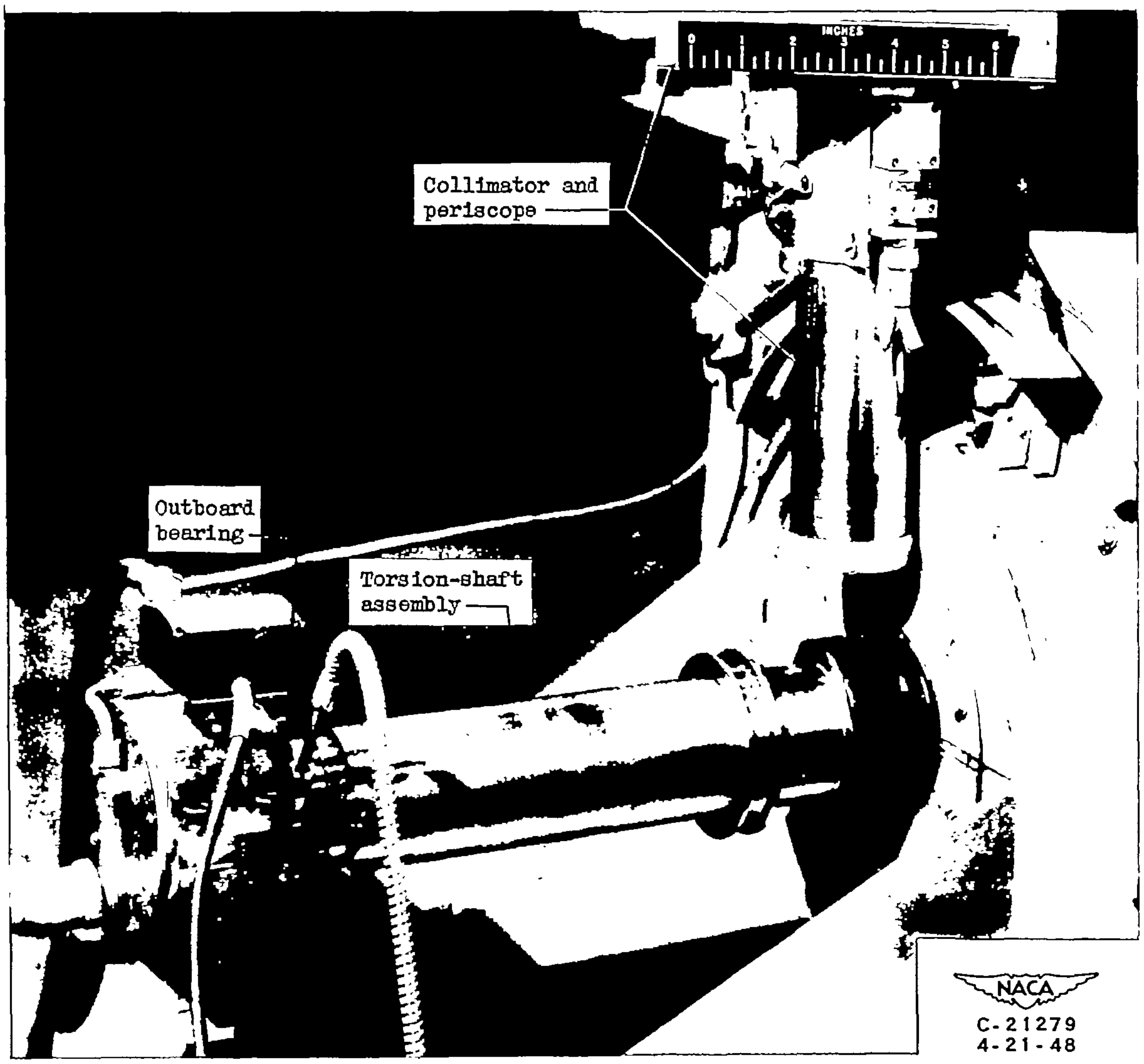


(a) Adaptor shaft and bearing for no-load spin test.



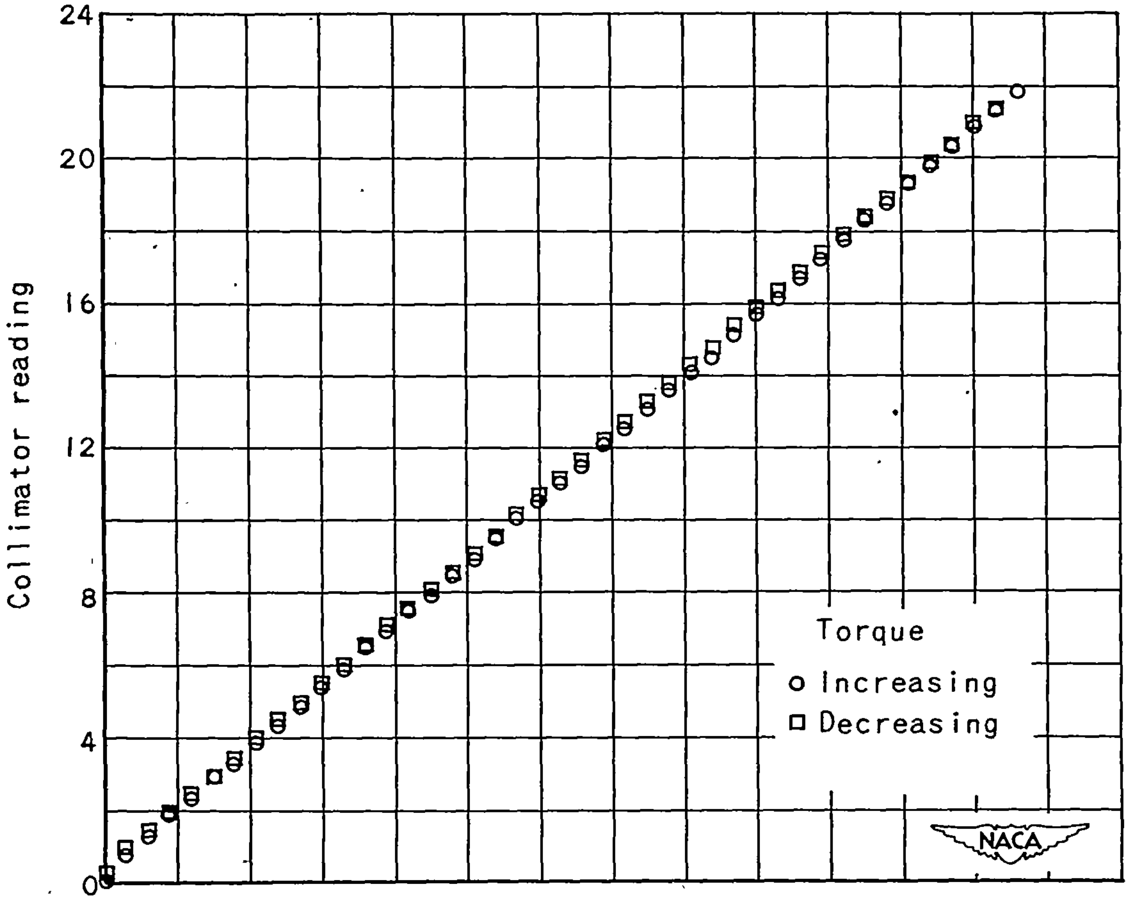
(b) Assembly of torsion and adaptor shaft for no-load spin test.

Figure 7. - Apparatus for no-load spin test.

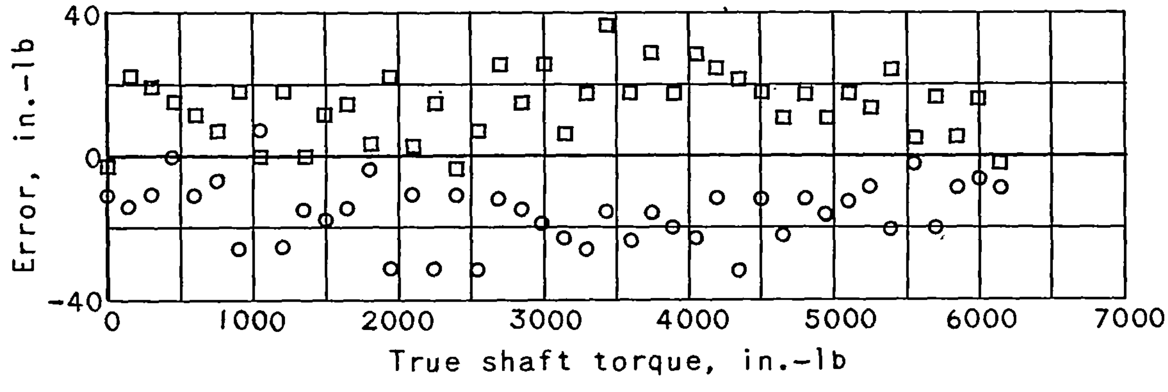


(c) Installation of torsion shaft for no-load spin test.

Figure 7. - Concluded. Apparatus for no-load spin test.



(a) Static calibration of torsion shaft.



(b) Error.

Figure 8. - Torsion-shaft static calibration.

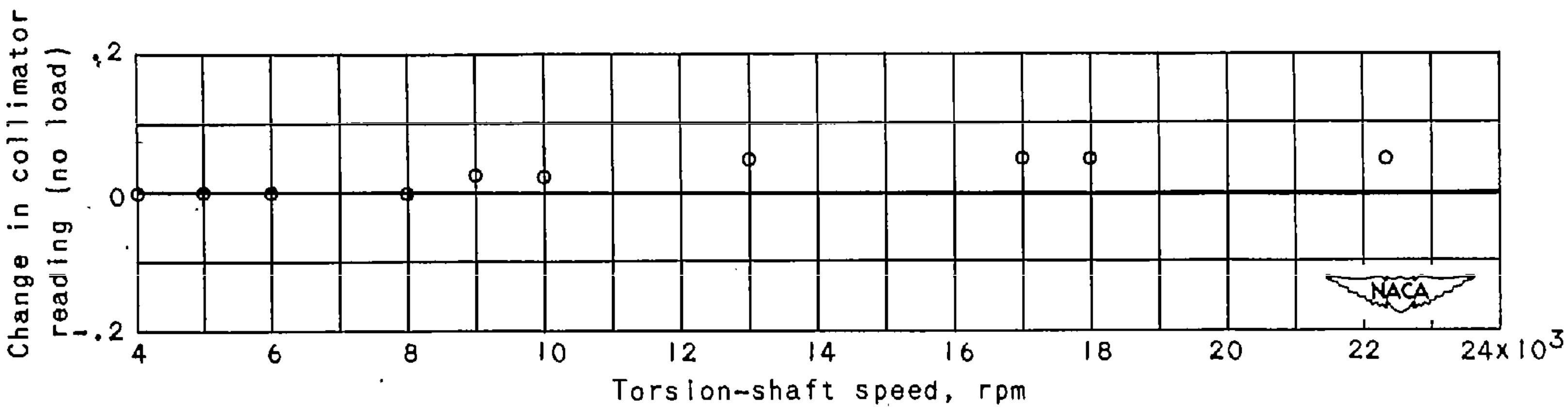


Figure 9. - Results of no-load spin tests.

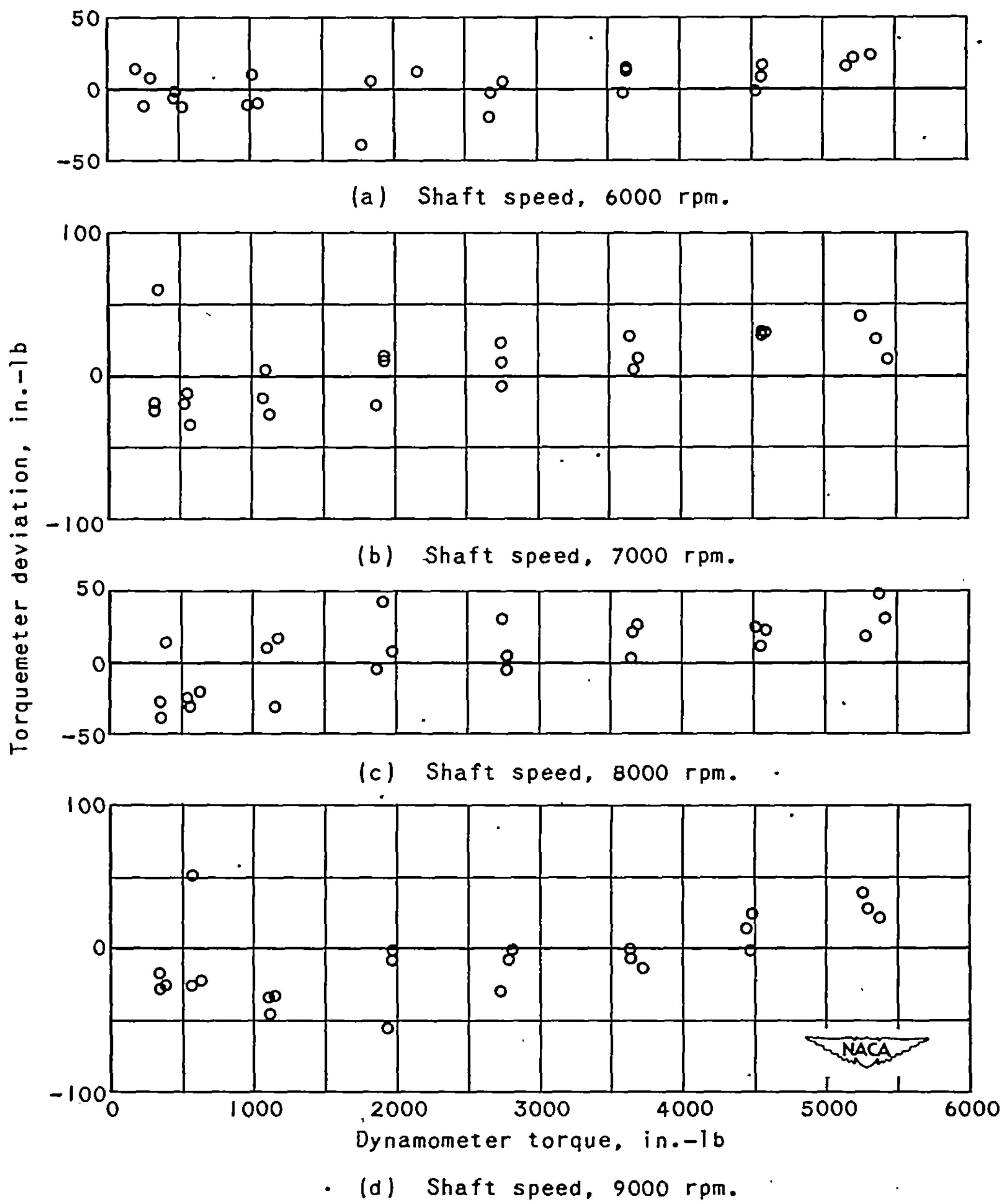
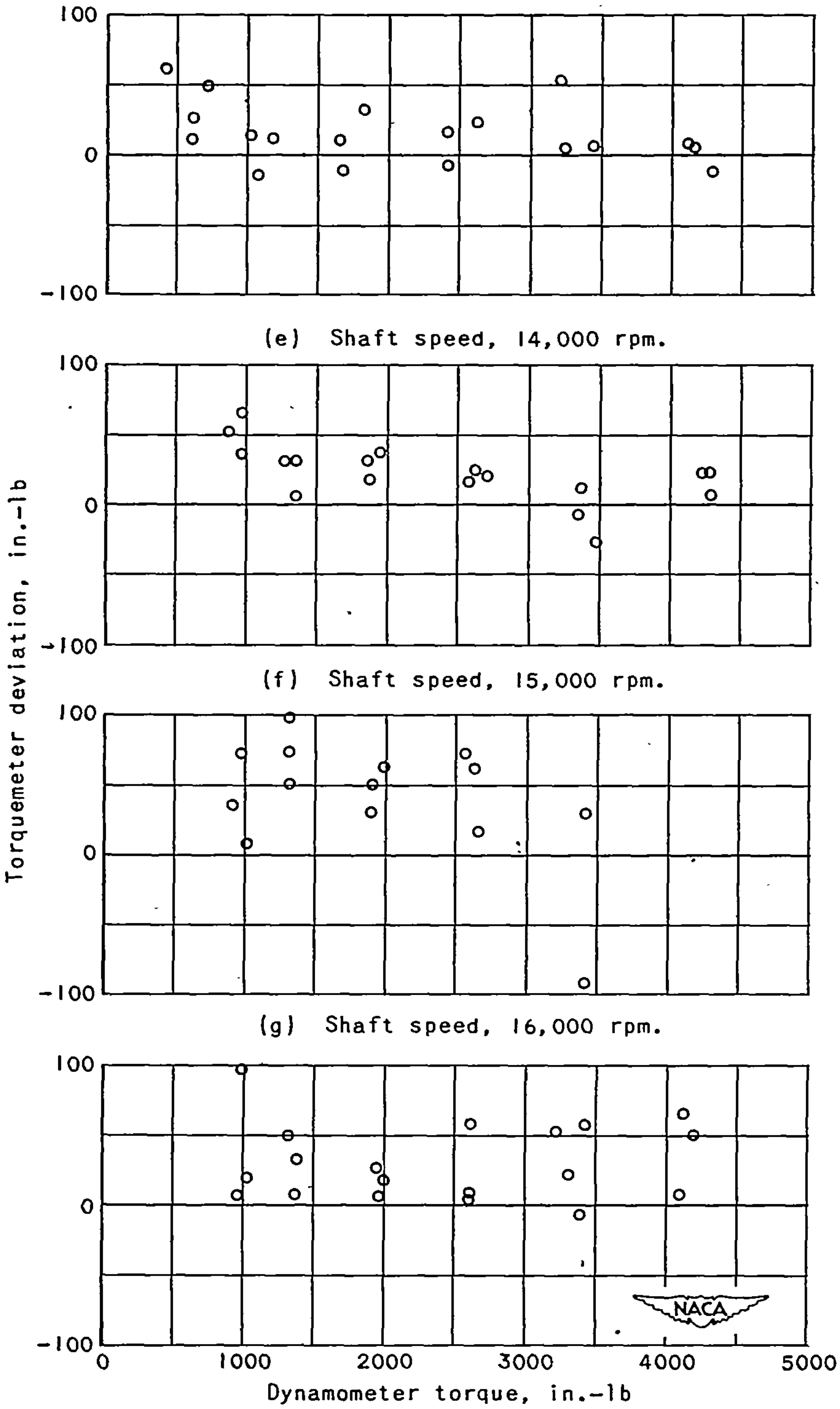


Figure 10. - Variation of torquemeter deviation for range of torque and speed.



(e) Shaft speed, 14,000 rpm.

(f) Shaft speed, 15,000 rpm.

(g) Shaft speed, 16,000 rpm.

(h) Shaft speed, 17,000 rpm.

Figure 10. - Concluded. Variation of torquemeter deviation for range of torque and speed.